E-4289

PERFORMANCE OF INERTED LUBRICATION SYSTEMS FOR TURBINE ENGINES

By R. L. Johnson, W. R. Loomis, and L. P. Ludwig

CONTENTS (For review copy only)

E-4289

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

TECHNICAL NOTE

PERFORMANCE OF INERTED LUBRICATION SYSTEMS FOR TURBINE ENGINES

By R. L. Johnson, W. R. Loomis, and L. P. Ludwig

ABSTRACT

Inerted lubricating systems containing 125 MM ball bearings and 6.33 inch diameter face contact seals were operated in simulated turbine engine sumps at speeds to 14000 RPM and temperatures to 800° F. The ball bearings operated satisfactorily to 600° F under 3280 pounds thrust load with 4 of the 5 lubricants evaluated. A persistent problem encountered was wear and leakage of the shaft seals. Additional experimental studies and analysis identified seal thermal deformation as a major factor in seal wear and leakage. New seals, revised to mitigate thermal deformations, were designed, analyzed and subjected to preliminary experimental studies.

E-4289

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

TECHNICAL NOTE

PERFORMANCE OF INERTED LUBRICATION SYSTEMS FOR TURBINE ENGINES

By R. L. Johnson, W. R. Loomis, and L. P. Ludwig

SUMMARY

Inerted lubricating systems containing 125 MM ball bearings and 6.33 inch diameter face contact seals were operated in simulated turbine engine sumps to 14000 RPM. Three hour screening tests, using degassed lubricants revealed that a dibasic acid ester (Mil-L-7808E) was not suitable for operation at 600° F because of lubricant related bearing failures. Three lubricants, an improved ester, a synthetic paraffin and a perfluorinated polymeric fluid performed satisfactorily at 700° F; the bearings were in excellent condition. A C-ether lubricant performed well at 600° F both with and without nitrogen inerting.

In inerted lubrication system operation the most troublesome component was the face contact seal (with bellows secondary) separating the nitrogen gas and oil. Analysis revealed that these seal

malfunctions (gas leakage and wear) were not related to the inerting gas but rather to seal thermal deformation. Further experiments, on contact and hydrostatic type seals in another simulated engine sump without inerting showed that thermal deformations were a major factor in limiting seal performance.

Two seal concepts revised to minimize thermal gradients and employing hydrodynamic lift devices were designed, analyzed and the lift concepts were experimentally checked. One revised design employed a hydrodynamic gas bearing for lift and the other obtained lift by means of an oil lubricated spiral groove bearing. Both of these revised designs showed low leakage potential in preliminary tests.

INTRODUCTION

Continuing increases in flight speeds and turbine inlet temperatures are raising the bulk temperatures of lubricating systems in aircraft turbine engines (refs. 1 and 2). The ester base synthetic lubricants of conventional lubricating systems have little or no margin to cope with higher temperatures (ref. 2). Therefore, not only is there an immediate need for improved lubricants and lubricating systems in uprated engines, but advanced engines, such as for Mach 3 flight, pose a much larger lubrication system problem because bulk lubricant temperatures are expected to be in the 450° to 500° F range.

In seeking solutions to this general problem of high temperature lubrication, some attention has been given to unconventional systems such as powder lubrication (ref. 3), dry films (ref. 4) and throwaway schemes. Another unconventional approach is based on oxygen exclusion through the use of an inert gas blanket (nitrogen). This inerted lubrication system approach is attractive because it may permit the use of presently available lubricants at significantly higher temperature levels. A key factor in the use of inerted lubricant systems is seal leakage, because the amount of blanket gas inventory (nitrogen) depends on leakage rates. This leakage consideration precludes the used of labeyrinth seals and places stringent requirements on the seal design since long life and low leakage must both be achieved. There is some evidence (ref. 5) that seal carbon life will be enhanced through the use of nitrogen inerting. This data, of reference 5, shows that carbon wear rate is lower in nitrogen and that oxidation is the chief cause of higher wear rates in air.

The objectives of this study were: (1) To determine the problems associated with operation of lubricants, bearing and seals in a full size simulated engine sump at speeds, pressures and temperatures expected in advanced engines; (2) To determine the leakage rates which can be expected of full size seals operating in an inerted bearing sump system; (3) To investigate newer seal concepts and to experimentable check the feasibility of these concepts.

The simulated engine bearing sump of the inerted lubrication system contained a 125 MM ball bearing operating at 14000 RPM with a 3280 pound thrust load. Runs were made with bearing temperatures between 600° and 800° F and with 100 psi pressure differential across the nitrogen gas to oil seal.

Another simulated engine bearing sump operating with air (not inerted) was used to evaluate seal concepts. This system contained bearings of the type presently used in turbine engines; a 150 MM roller bearing and a 100 MM duplex thrust bearing. The air to oil seals were operated at pressure differentials to 300 psi and at speeds to 400 ft/sec. Sealed air temperatures to 1200° F were employed.

Analytical studies were made to determine thermal gradients in the seal structure and to determine overall elastic deformation due to temperature, pressure and centrifugal force.

Part of the studies reported herein were made under NASA contract NAS 3-7609 (ref. 6) and under NASA contract NAS 3-6267 (ref. 7).

APPARATUS AND PROCEDURE

Inerted Lubrication Systems

A schematic of the simulated turbine engine sump employing an inerted lubricating system is shown in figure 1. The rig contained a 125 MM ball bearing and was operated at 14000 RPM. Heaters on the bearing and housing O.D. permitted operation to 800° F bearing outer race temperature. Lubricants were introduced at temperatures to 500° F. Both recirculation and mist systems were employed. Two face contact seals (bellows secondary) formed the sealing system. Nitrogen was introduced between the 2 seals at 105 psi. The seal between the nitrogen gas and lubricant, therefore, was subjected to 105 psi pressure differential, and the seal between the hot air (1200° F and 100 psi) and nitrogen was subjected to 5 psi pressure differential.

Figure 2 is a schematic of the seal employed in the inerted lubrication systems. The bellows assembly was fabricated from inconel and a finger spring damper, rubbing against the end piece O.D., provided friction damping. The carbon-graphite nosepiece face contained three elements, an outer wear pad, a sealing dam and an inner wear pad. The wear pads were interrupted by grooves which vented the wear pad area. Thus the pressure drop occurred only across the sealing dam. The seal seat was chrome plated on the rubbing surface and flat within 3 light bands.

Seal gas leakage was continously monitored and inspection after running provided wear measurements of the carbon-graphite nosepiece and seal seat. In some cases surface profile traces were made to determine effect of sealing face deformation on contact area. Lubricant coking observations were also made. Other parameters recorded included sealed gas pressure and temperature, sliding speed, lubricant temperature in and out and bearing outer race temperature.

The test bearing was a split inner ring, angular-contact ball bearing; the type most widely used in aircraft propulsion turbine. This design permits a maximum ball compliment (because of separable inner ring halves) and supports thrust load in either direction. The separable ring also permits the use of a precision-machined one-piece cage which is required for high-speed high-temperature operation. The test bearings have a bore diameter of 125 MM and a nominal mounted operating contact angle of 26°. This bearing runs at the test speed of 14000 RPM ($dn = 1.75x10^6$) and a thrust load P = 3280 lbs. For operating temperatures up to 600° F consumable electrode vaccum melted (CVM) M-50 tool steel rings and balls were used. At higher temperatures CVM WB49 tool steel were used for the bearing rings and CVM M-1 tool steel for the balls. The cages are of an outer-ring piloted design and were constructed of silver plated M-1 tool steel. The bearings had nominal 51.6% inner ring conformity, a 52.1% outer ring conformity, 4 micro-inch RMS maximum across grooves, 21-13/16 diameter balls and .0068-.0080 inch unmounted internal radial looseness.

The lubricants used were:

(a) Dibasic acid ester (Mil-L-7808E type) which is a mono-hydric alcohol formulated with proprietory additives. The viscosity extrapolated to 600° F is 0.64 cs.

(b) Ester-base lubricant (MIL-L-7808E type) with improved thermal stability and estimated 1.17 cs at 600° F.

(c) Synthetic paraffinic lubricant containing a proprietory boundary lubricant additive. The viscosity is an estimated 2.4 cs at 600° F.

(d) Perfluorinated polymeric (fluorocarbon).

(e) C-ether (modified polyhenyl Ether).

Seal Concept Studies

Figure 3 is a schematic of the seal and bearing area of a simulated turbine engine sump employing an open lubrication system and used to study face contact and hydrostatic seal concepts. The system simulated the roller bearing sump at the turbine location and the structure is typical of engine parts. High pressure air (to 1200° F) was introduced at the seal dam I.D. and air leakage was into the bearing compartment. Seal gas leakage was continously monitored and inspected after running provided wear measurements of the sealing faces. Other parameters recorded included sealed gas pressure and temperature, sliding speed, lubricant temperature and seal nosepiece temperature. In some runs miniature accelerometers were attached to the nosepiece and accelerometer output was recorded on magnetic type and then analyzed for evidence of nosepiece instability.

Analysis of sealing face deformation was made by first calculating a thermal map of the seal assembly by finite difference steady state heat transfer program (computer). Thermal deformation was generated from the thermal map by axi-symmetric finite element program (computer) which also included pressure and centrifugal force effects.

RESULTS AND DISCUSSION

Inerted Lubrication System

Operation of a simulated lubrication system (bearings, seals and pumps) allows evaluation of a lubricant at temperatures, shear rates, and loads which are indicative of actual operation. Thus a check is obtained on such items as coking, lubricant breakdown due to shear rates, lubricant effectiveness at bearing cage sliding surface, corrosion, seal performance, etc. The lubrication system, which is described in the apparatus section, simulated the expected environmental conditions of an inerted bearing sump of an advanced engine. The lubricants evaluated, also described in the apparatus section, were: (1) a dibasic acid ester qualifying against MIL-L-7808E, (2) an improved ester similar to MIL-L-7808E, (3) a synthetic paraffinic lubricant, (4) a C-ether (modified polyhenyl ether, and (5) a perfluorinated polymeric oil. All lubricants were degassed before use.

Primary results obtained for these lubricants in the system studies (3-hour screening tests) are as follows:

(a) The Mil-L-7808E type lubricant was determined to be not suitable, even with inert blanketing, at 600° F outer race bearing temperatures and above because of lubrication-related bearing failures.

(b) An improved ester (similar to Mil-L-7808E type) of somewhat greater vescosity than the useful Mil-L-7808 E ran successfully in tests up to 750° F. At 650° F bearing temperature, this fluid performed satisfactorily for approximately ten hours before testing was stopped due to test seal malfunctioning.

(c) The synthetic paraffinic lubricant was tested satisfactorily at temperatures up to and including 700° F. An attempted runs at 750° F was aborted after less than two hours because of excessive leakage of the oil-side test seal.

(d) The C-ther performed satisfactorily at 600° F both with and without nitrogen blanketing. Higher temperature testing was suspended due to apparent bearing thermal instability with this lubricant.

(e) Perfluorinated polymeric fluid was tested successfully to temperatures of 700° F but higher oil flow rates were required than for the improved ester or the synthetic paraffinic lubricants.

Only minor oil coking occurred in most test and was not to the extent to seriously affect bearing or seal performance. Longer term tests of 50 hours with the improved ester and the synthetic paraffinic were not successful due to repeated oil-side seal malfunctions, which was also the limiting factor in a majority of the three hour screening tests.

From the overall system viewpoint, reliable bearing and seal operation using inerted recirculating lubrication appears to be feasible at 150 to 200° F higher temperatures than possible with a conventional recirculating system with several off-the-shelf fluids. The primary problem is related to achieving oil-seal performance to reduce loss of inerting gas.

Analysis revealed that the seal malfunctions were not related to the inerting gas but rather to thermal deformation. Seal operational leakage data is summarized in table I. The data shows that seal leakage, for the majority of runs, is greater than would likely be acceptable from a nitrogen inventory standpoint, greater than 5 SCFM. The leakage shown is a total for both seals (oil seal and air seal of fig. 1) in the system, however, it was determined that most of the leakage (up to 90%) could be attributed to the oil seal.

One seal malfunction which commonly occurred was a cyclic increase and decrease in leakage. This cyclic change in leakage implies a cyclic change in the average height of the sealing gap (see fig. 2). And a mechanism which could account of this cyclic malfunction starts in the initial operation in which the closing force, being slightly larger than the opening force, tends to hold the nosepiece in sliding contact against the seal seats (see fig. 2). However, a small average dynamic gap exists as is evidenced by the seal leakage (see fig. 4 for calculated leakage as a function of gap height). But the heat generation at the sliding interface is relatively high because of sliding contact and shearing of a relatively thin film of gas. This high heat generation causes more thermal growth in the nosepiece than in the bellows. The opening force, therefore, increases with respect to the closing force and the seal eventually opens. Now with increased seal gap the heat generated is reduced, the nosepiece cools and the seal returns to the initial condition and the cycle repeats itself.

Coke deposits from lubrication degradation were evident along the outer wear pad after some of the runs (see fig. 2 for location of wear pads). This coking was attributed to the heat generated at the sliding face of the carbon, thus the area adjacent to the dam is hotter than the rest of the nosepiece. In several cases the coking was severe enough to plug the outer wear pad vents (see fig. 2 for vent locations). It should be noted that plugging of these wear pad vents can also cause seal lift and attendent high leakage.

Another problem encountered was the bellows closing-force area decrease which accompanied the pressure increase. This is sometimes called a change in bellows mean effective diameter (m.e.d.) (See fig. 2). Balance of the opening and closing forces requires a knowledge of this m.e.d. change and of the probable pressure profiles at the sealing dam. At best the selected force balance was a compromise and a slight closing force bias was selected at 105 psi in order to preclude opening of the sliding interface (dam) due to inertia forces. Carbon wear encountered in some of the tests was attributed to the lack of control over this seal force balance.

Evaluation of Seal Concepts In an Open Lubrication System

Face contact and hydrostatic seal concepts were evaluated in a simulated engine sump operating with an air environment (not inerted). The face contact seal used operates on the same principle as the seal previously described for the inerted system except that the bellows is replaced with 2 piston rings and a series of helical springs (fig. 5). It is evident from table II, which is a summary of the test data, that seal leakage was the primary failure mode. Additional data characteristic of the leakage rates of these seals is given in figure 5. Up to 120 psi pressure differential the leakage rate is relatively low (being less than 5 SCFM), but beyond 120 psi the leakage rate shows a strong dependence on sliding speed. This dependence of leakage on sliding speed could be caused by inertia effects (ei nosepiece dynamic response to seat runout and rotation) or thermal effects such as described for the bellows face seal.

An additional problem associated with operation of the face contact seal was the thermal deformation of the nosepiece and seal seat. This thermal deformation is illustrated in figure 6. The axial thermal gradients cause both the nosepiece and seal seat to form divergent leakage gaps. And a diverging gap reduces the seal opening force and causes the net closing force increase. The result is heavy carbon wear with the I.D. showing more wear because of the deformation.

A second concept evaluated was an orifice-compensated hydrostatic seal design to operate on an air film of about .0005 inches. The seal design (fig. 7) is somewhat similar to that of a face contact seal except that a recess and a series of orifices have been added to the nosepiece face. Some of the leakage through the seal takes place through the orifices aranged circumferentially around the seal. The leakage flow through the orifices produces a pressure drop from the sealed pressure P_1 to the recess pressure P^1 and the sealing gap height is controlled by compensations in recess pressure P^1. The mechanism works like this; if the gap is closed down for some reason, the leakage out is reduced. This means low pressure drop in the orifices due to reduced leakage flow; therefore, recess pressure P' will increase, approaching sealed pressure P_1, and producing a net restoring force to maintain design gap height. Similarly, if the gap opens beyond the design point, the inverse process takes place.

Table III contains a summary of pertinent results obtain in operation of the orifice-compensated hydorstatic seal. Inspection of the data revealed that the most serious problems were excessive leakage and excessive rubbing due to thermal deformation. The effects of thermal deformation are shown in figure 9 which is data taken at 200 feet per second rubbing speed and 100 psi pressure differential. With a sealed air temperature of 120° F the seal leakage is near 13 SCFM and as the air temperature is increased the leakage decreases. This leakage decrease is due to increasing angular deformation of the sealing gap. The net result is that the seal runs with closer clearance. Eventually, as the air temperature increased, the nosepiece rubbed against the seal seat and failure occured.

The preceding data points to thermal deformation as being a serious problem in all three seals evaluated (face contact with bellows, face contact with piston ring and orifice-compensated seals). Accordingly an analysis was made of means to reduce thermal deformation in the seal structure and analysis revealed that the deformation causing divergent leakage gaps at the sliding interface was due to: (a) axial thermal gradients in the seat and nosepiece, (b) non-uniform thermal growth of the shaft under the seal, and (c) moments induced by the thermal growth of the spacers clamping the seal seat. The other serious problem encountered in the seal experimental evaluations was wear due to high speed rubs. Therefore, the revised seal designs contained provisions for mitigating thermal deformation and hydrodynamic devices to prevent contact between the nosepiece and the seal seat. The 2 revised designs are shown in figures 9 and 10.

Face Contact Seal with Gas Bearing for Hydrodynamic Lift

The revised face contact seal design (fig. 9) consists of a structually isolated seal seat which is mormted over its centroid on a shaft spacer with radial flexibility. The seal seat is clamped axially by a machined bellows. Oil is passed under the shaft spacer, thus aiding thermal isolation, and then thru radial holes in the seal seat near the rubbing interface. Molybdenum was selected for the seat because it has a low thermal deformation factor; ei high thermal conductivity and low thermal expansion (ref. 8). The nosepiece and nosepiece carrier are both piloted by 3 locating lugs which also serve as anti rotation devices. Thin shields under the piston carrier and over the shaft provides additional thermal shielding. The oil which passes through the rotating seal seat is caught by a baffle and re-directed back to cool the nosepiece and nosepiece carrier. The nose-piece face contains 3 elements: (a) a pad type gas bearing, (b) a sealing dam which acts as a conventional face contact seal and (c) a spiral groove windback section of large axial clearance (.020 inch) which prevents oil from seeping into the sealing dam area. The windback section is not affected by thermal deformation because of this large axial clearance.

The thermal map for the nosepiece of this revised design is given in figure 11, and 12 contains the combined deformation due to thermal pressure effects. The analysis at 300 psi and 1300° F gas temperature shows that angular deformation of the nosepiece gas bearing section is 1.3 milli radians. This is not serious since the calculated film height would then be 0.00020 inch at the bearing I.D. and 0.00050 at the gas bearing O.D., ei the gas bearing can accomodate this tilt angle. Of more concern is the milli radian deformation across the sealing dam.

Since this deformation has a divergent tendency, the opening force is expected to decrease with increasing deformation; however, the analysis shows that the gas bearing will accept the force changes without touchdown. Figure 13 shows the calculated gas bearing load capacity.

Preliminary runs were made with hydrodynamic type seals having a gas bearing pad type geometry incorporated into the nosepiece (similar to fig. 9). The purpose of the runs was to check the hydrodynamic concept and the runs were made with room temperature air. Inspection of the seal surfaces indicated lift was occurring, however a seal failure was experienced at 400 feet per second and 100 psi. Seal leakage measured is given in figure 14. Excellent sealing potential is shown by the low leakage obtained to 300 feet per second and 200 psi.

Face Contact Seal with Lubricated Spiral Groove for Hydrodynamic Lift

A second design for providing hydrodynamic lift is shown schematically in figure 10. The overall design is structurally similar to the previous seal except that the gas bearing pads have been eliminated and seat cooling oil is made to pass through a spiral groove section which provides hydrodynamic lift (See ref. 9 for spiral groove seal theory). In operation, the spiral groove section provides the lift to establish a sealing gap of about .0005 inches. This lift prevents wear, limits the gas leakage to acceptable levels and mitigates the effects of thermal angular deformation. Preliminary runs with seals employing this spiral lift concept showed operation at less than 0.0005 inch film thickness and .5 SCFM leakage at 100 psi ft/sec sliding velocity. Typical leakage data is given in figure 14.

CONCLUDING REMARKS

Nitrogen gas inerted lubricated system studies were made using a simulated engine sump operating at 14000 RPM with 125 MM ball bearing (3260 lbs thrust) and 6.33 inch mean diameter face contact seals. Three hours screening runs were used to evaluate candidate lubricants to 800° F bearing temperatures. Seal concept studies were also made in open lubrication systems. Face contact and hydrostatic seal performance was determined at various speeds, pressures and temperatures. Design revisions were made to seals in order to mitigate problems encountered.

The experimental data and analysis revealed the following:

1. Dibasic acid ester (Mil-L-7808E) is not suitable for 600° F inerted operation because of lubrication related bearing failures. Three lubricants, an improved ester (twice the viloscity of MIL-L-7808E), a synthetic paraffinic fluid, and a C-ether performed satisfactorily at 600° F bearing temperature. Perfluorinated polymeric fluid performed satisfactorily at 600° F however there was slight evidence of corrosion attack on the bearing.

2. The inerted system operated satisfactorily except for the bellows face contact seal which had high leakage and wear. This high leakage and wearwas attributed to thermal deformation of the sealing force.

3. Additional seal studies (in open lubrication systems) on face contact and on hydrostatic seals, also revealed that seal face thermal deformation was a major problem.. Changes in force balance, which accompany thermal deformation, lead to severe rubbing and wear.

4. Two face seal concepts, one using a hydrodynamic gas bearing for lift, and the other a oil lubricated spiral groove for lift, showed low leakage in preliminary dynamic testing.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio.

Alt/1-16-68

E-4289

REFERENCES

1. Bisson, E. E., and Anderson, W. J.: Advanced Bearing Technology NASA SP-38.

2. Reynolds, H. W. Jr.: Turbine Oil and Test Method Development Needed for Current and Future Engines. Proceedings of the USAF-SW RI Turbine Engine Lubrication Conference (Sept. 13-15, 1966).

3. Schlosser, A. L.: Powder Lubrication of Rolling Contact Bearings at Very High Speeds and Temperatures. Proceedings of the USAF Airospace Fluids and Lubricants Conference, San Antonio, Texas April, 1963.

4. Sliney, H. E.: Bearing Run at 1250° F with Solid Lubricants - "Space Aeronautics", Vol. 35, No. 3, March 1961.

5. Strom, T. N., Allen, G. P., and Johnson, R. L.: Wear and Friction of Impregnated Mechanical Carbons at Temperatures to 1400° F (760° C) in Air and Nitrogen. NASA TN D-3958, May 1967.

6. Parks, A. J., McKibben, R. H., and Ng, S. S. W.: Development of Main Shaft (NASA Contract NAS 3-7609).

7. Rhoads, W. L., and Sibley, L. B.: Supersonic Transport Lubrication System Investigation. SKF report No's CR 54311, CR 54312, and CR 54313.

8. Ludwig, L. P., Strom, T. N., Allen, G. P., and Johnson, R. L.: Improving Performance of Face Contact Seals in Liquid Sodium (400° to 1000° F) by Incorporation of the Spiral-Groove Geometry. NASA TN D-3942, May 1967.

9. Strom, I. N., Ludwig, L. P., Allen, G. P., and Johnson, R. L.: Spiral Groove Face Seal Concepts; Comparison to Conventional Face Contact Seals in Sealing Liquid Sodium (400° to 1000° F). ASME paper 67 WA/Lub - 17.